Chapter 5
Machinery Installation, Propulsion and Steering Devices

Abstract The study of a ship's machinery installation and her propulsion plant, namely, of the main machinery (prime mover) and auxiliary engines, of the propeller and the required rudder, is a subject of specialized literature, a sample of which is cited in the given list of references. In the context of the present book, we will limit ourselves to some general comments on the selection criteria and the recommended procedure regarding the selection of ship's machinery installation and of the propulsive and steering devices as to meet the needs of the preliminary design of a ship.

5.1 Selection of Main Machinery

A. Selection Criteria

1. Acquisition cost
2. Safety in operation
3. Weight
4. Space requirements
5. Specific fuel oil consumption (SFOC)
6. Fuel type and fuel cost
7. Emission of toxic gases (SO_X, NO_X etc.)
8. Repair cost
9. Maintenance cost
10. Manoeuvrability
11. Vibrations
12. Automation of control

The above selection criteria for a ship's main engine installation are valid regardless of the specific conditions, which may apply to a particular ship. However, it is characteristic that certain aspects and selection criteria were strongly emphasized in the past, such as in the early 1970s, the introduction of automated systems in ship engine installations. Also, after the first oil crisis in the early 1970s and the rapid increase of crude (and fuel) oil price, the specific fuel oil consumption, the type of fuel and the savings of energy in general play a primary role in the selection of the appropriate main propulsion engine of a ship.

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More recently (especially after 2008), following relevant discussions at the United Nations Organization (UNO) regarding the protection of the air environment from toxic gases emitted by engines of all types of transportation vehicles, the International Maritime Organization (IMO) proceeded introducing a framework of regulations for the reduction of gaseous pollutants (CO_2, NO_X, SO_X) of marine engines, so as to define upper limits for the emissions of gaseous pollutants for ships. In this respect, an Energy Efficiency Design Index (EEDI)[1] has been introduced for most types of merchant ships, which needs to be kept below a certain limiting value that is specific to the ship's type and size.

The EEDI of a ship is a measure of the ship's energy efficiency and her green house gas (GHG) emission level, expressed in [grams CO_2 per tonne mile]; it is calculated by the following formula (see MEPC 212(63) 2012):

$$\mathrm{EEDI} = \frac{\left(\prod_{j=1}^{n} f_j\right)\left(\sum_{i=1}^{n\mathrm{ME}} P_{\mathrm{ME}(i)} \cdot C_{\mathrm{FME}(i)} \cdot SFC_{\mathrm{ME}(i)}\right) + \left(P_{\mathrm{AE}} \cdot C_{\mathrm{FAE}} \cdot SFC^{*}_{\mathrm{AE}}\right) + \left(\left(\prod_{j=1}^{n} f_j \cdot \sum_{j=1}^{n\mathrm{PTI}} P_{\mathrm{PTI}(i)} - \sum_{i=1}^{n\mathrm{eff}} f_{\mathrm{eff}(i)} \cdot P_{\mathrm{AEeff}(i)}\right) C_{\mathrm{FAE}} \cdot SFC_{\mathrm{AE}}\right) - \left(\sum_{i=1}^{n\mathrm{eff}} f_{\mathrm{eff}(i)} \cdot P_{\mathrm{eff}(i)} \cdot C_{\mathrm{FME}} \cdot SFC^{**}_{\mathrm{ME}}\right)}{f_i \cdot f_c \cdot Capacity \cdot f_w \cdot V_{\mathrm{ref}}} \tag{5.1}$$

where:

C_F is a non-dimensional conversion factor between fuel consumption measured in gram and CO_2 emission also measured in gram based on carbon content. The subscripts ME(i) and AE(i) refer to the main and auxiliary engine(s) respectively. C_F corresponds to the fuel used when determining SFC listed in the applicable test report included in a Technical File as defined in paragraph 1.3.15 of NO_x Technical Code. The value of C_F as follows (in Table 5.1):

V_{ref} is the ship speed, measured in nautical miles per hour (knot), on deep water in the condition corresponding to the Capacity as defined below (in case of passenger ships and ro-ro passenger ships, this condition should be summer load draught) at the shaft power of the engine(s) and assuming the weather is calm with no wind and no waves.

Capacity is defined as follows:

- For bulk carriers, tankers, gas tankers, ro-ro cargo ships, general cargo ships, refrigerated cargo carrier and combination carriers, deadweight should be used as *Capacity*.
- For container ships, 70 % of the deadweight (DWT) should be used as Capacity.

[1] The EEDI was made mandatory for *new* ships, as of January 1, 2013; this was decided at MEPC 62 (July 2011) with the adoption of amendments to MARPOL Annex VI (resolution MEPC.203(62)) and went along with the introduction of a Ship Energy Efficiency Management Plan (SEEMP) for *all ships* (see, also Chap. 1 of this book).

Table 5.1 C_F values

Type of fuel	Reference	Carbon content	C_F (t-CO2/t-fuel)
1. Diesel/gas oil	ISO 8217 Grades DMX through DMB	0.8744	3.206
2. Light fuel oil (LFO)	ISO 8217 Grades RMA through RMD	0.8594	3.151
3. Heavy fuel oil (HFO)	ISO 8217 Grades RME through RMK	0.8493	3.114
4. Liquified petroleum gas (LPG)	Propane	0.8182	3.000
	Butane	0.8264	3.030
5. Liquified natural gas (LNG)		0.7500	2.750

Deadweight means the difference in tonnes between the displacement of a ship in water of relative density of 1,025 kg/m^3 at the summer load draught and the lightweight of the ship. The summer load draught should be taken as the maximum summer draught as certified in the stability booklet approved by the administration or an organization recognized by it.

P is the power of the main and auxiliary engines, measured in kilowatt. The subscripts ME and AE refer to the main and auxiliary engine(s), respectively. The summation on *i* is for all engines with the number of engines (*n*ME).

V_{ref},*Capacity* and *P* should be consistent with each other.

SFC is the certified specific fuel consumption, measured in gram per kilowatt-hour, of the engines. The subscripts ME(*i*) and AE(*i*) refer to the main and auxiliary engine(s), respectively. The SFC should be corrected to the value corresponding to the ISO standard reference conditions using the standard lower calorific value of the fuel oil (42,700 kJ/kg), referring to ISO 15550:2002 and ISO 3046-1:2002.

f_j is a correction factor to account for ship specific design elements.

f_w is a non-dimensional coefficient indicating the decrease of speed in representative sea conditions of wave height, wave frequency and wind speed (e.g. Beaufort Scale 6).

$f_{eff(i)}$ is the availability factor of each innovative energy efficiency technology.

$f_{eff(i)}$ for waste energy recovery system should be one (=1.0).

f_i is the capacity factor for any technical/regulatory limitation on capacity, and should be assumed to be one (1.0) if no necessity of the factor is granted.

f_c is the cubic capacity correction factor and should be assumed to be one (=1.0) if no necessity of the factor is granted.

Length between perpendiculars, Lpp means 96 % of the total length on a waterline at 85 % of the least moulded depth measured from the top of the keel, or the length from the foreside of the stem to the axis of the rudder stock on that waterline, if that were greater. In ships designed with a rake of keel the waterline on which this length is measured should be parallel to the designed waterline. Lpp should be measured in metre.

According to Regulation 20 of Annex VI of Chapter 4 IMO MARPOL 73/78 (IMO MEPC 203(62) 2011), the attained EEDI shall be calculated for each new ship, or

Table 5.2 Reduction factors (in %) for the EEDI relative to the EEDI reference line

Ship type	Size	Phase 0 1 Jan 2013–31 Dec 2014	Phase 1 1 Jan 2015–31 Dec 2019	Phase 2 1 Jan 2020–31 Dec 2024	Phase 3 1 Jan 2025 and onwards
Bulk carrier	20,000 DWT and above	0	10	20	30
	10,000–20,000 DWT	n/a	0–10*	0–20*	0–30*
Gas carrier	10,000 DWT and above	0	10	20	30
	2,000–10,000 DWT	n/a	0–10*	0–20*	0–30*
Tanker	20,000 DWT and above	0	10	20	30
	4,000–20,000 DWT	n/a	0–10*	0–20*	0–30*
Container ship	15,000 DWT and above	n/a	0–10*	0–20*	0–30*
	10000–15000 DWT	n/a	0–10*	0–20*	0–30*
General cargo ship	15,000 DWT and above	0	10	15	30
	3,000–15,000 DWT	n/a	0–10*	0–15*	0–30*
Refrigerated cargo ship	5,000 DWT and above	0	10	15	30
	3,000–15,000 DWT	n/a	0–10*	0–15*	0–30*
Combination carrier	20,000 DWT and above	0	10	20	30
	4,000–20,000 DWT	n/a	0–10*	0–20*	0–30*

* Reduction factor to be linearly interpolated between the two values dependent upon vessel size. The lower value of the reduction factor is to be applied to the smaller ship size.

any ship that has undergone a major conversion, which is so extensive that the ship is regarded by the Administration as a newly constructed ship which falls into one or more of the categories in regulations 2.25 to 2.35 (as shown in the below table). The attained EEDI shall be verified, based on the EEDI technical file, either by the administration or by any organization duly authorized by it.

According to Regulation 21 of Annex VI of Chapter 4 MARPOL 73/78, the attained EEDI shall be less/equal a required level, set by regulation, as follows:

$$\text{Attained EEDI} \leq \text{Required EEDI} = (1 - x)\ \text{Reference line value} \tag{5.2}$$

where x is the reduction factor specified in Table 5.2 for the required EEDI compared to the EEDI Reference line.

The reference line values shall be calculated as follows (IMO MEPC 215(63) 2012):

Table 5.3 Parameters for determination of reference values for the different ship types

Ship type defined in Regulation 2 of Annex VI of Chap. 1 MARPOL 73/78	*a*	*b* = *Capacity*	*c*
2.25 Bulk carrier	961.79	DWT	0.477
2.26 Gas carrier	1120.00	DWT	0.456
2.27 Tanker	1218.80	DWT	0.488
2.28 Container ship	174.22	DWT	0.201
2.29 General cargo ship	107.48	DWT	0.216
2.30 Refrigerated cargo carrier	227.01	DWT	0.244
2.31 Combination carrier	1219.00	DWT	0.488

$$\text{Reference line value} = a \times b^{-c} \tag{5.3}$$

where *a, b* and *c* are the parameters are given in Table 5.3.

The following figures represent typical reference lines for bulkcarriers, tankers and general cargo ships to be used in the assessment of EEDI according to the IMO MEPC 62/6/4 (2011): Consideration and adoption of amendments to mandatory instruments—Calculation of parameters for determination of EEDI reference values (Figs. 5.1, 5.2 and 5.3).

The following figure is from Lloyd's Register (2012): Implementing the EEDI—Guidance for owners, operators, shipyards and tank test organizations (Fig. 5.4):

The key measures for reducing gaseous toxic emissions from marine engines, which goes along with the reduction of fuel consumption, are as follows:

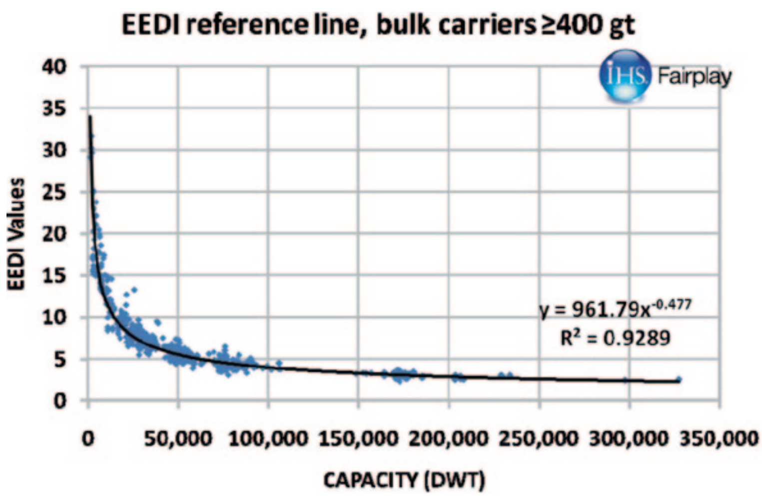

Fig. 5.1 Typical reference lines for bulk carriers (IMO MEPC 62/6/4 2011)

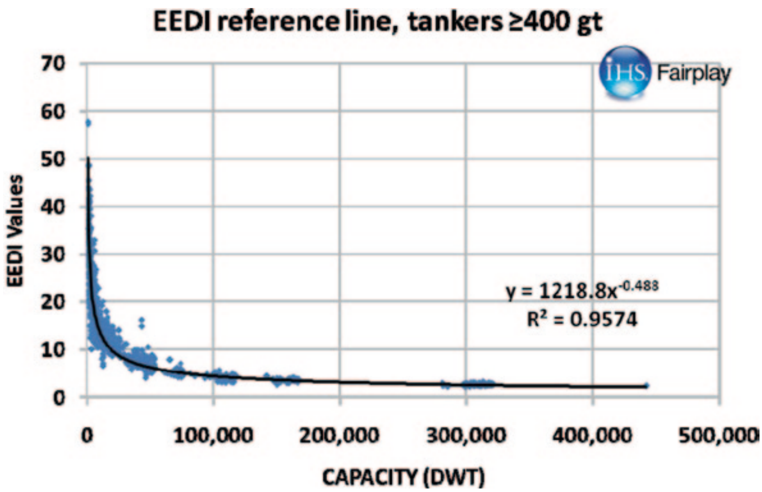

Fig. 5.2 Typical reference lines for tankers (IMO MEPC 62/6/4 2011)

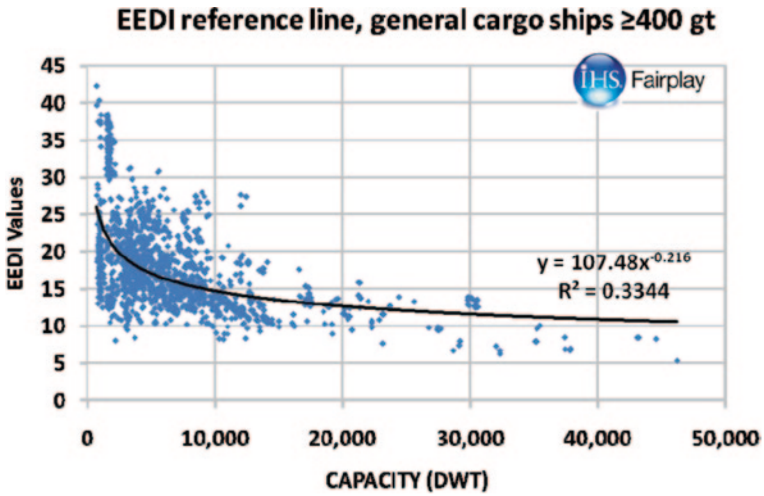

Fig. 5.3 Typical reference lines for general cargo ships (IMO MEPC 62/6/4 2011)

- Reduction of fuel consumption through reduction of ship's resistance and powering
  - Optimization of ship's hull form leading to a reduction of the required propulsion power for specified speed (calm water performance and added resistance in seaways, *new-buildings*)
  - Fitting of propulsive efficiency enhancing devices (stern flow ducts, spoilers, CPT propellers etc., for *existing ships and to some extent, new buildings*)

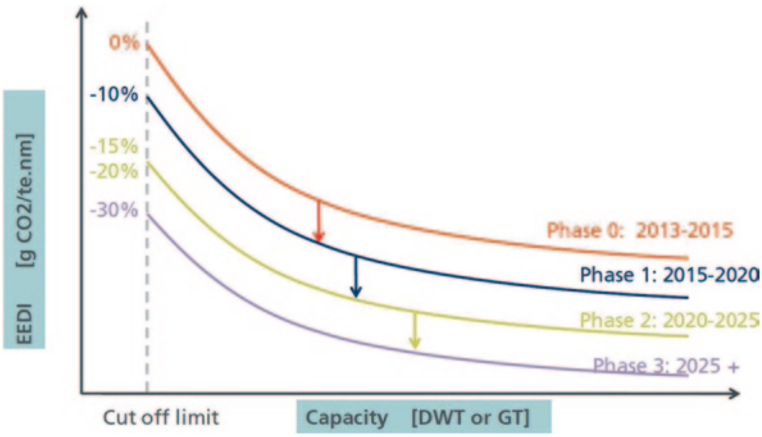

Fig. 5.4 EEDI concept (LR 2012)

- Refitting of bulbous bow (*existing ships*)
- Optimization of operational trim (*existing ships*)
- Minimization of the amount of carried ballast water (*new buildings and existing ships*)
- Reduction of viscous resistance through special treatment of wetted surface (paints etc.) and other innovation measures (release of air bubbles etc…) *(mainly new-buildings*)
- Optimization of ship's routeing
- Reduction of service speed (*slow steaming*)

• Improvement of marine engines' technology

- Reduction of SFOC
- Reduction of toxic gas emissions
- Dual fuel consumption (HFO/MDO–LNG)

• Improvement of fuel quality

- Introduction of bio-fuels for marine engines

Figure 5.5 below shows indicative values for the specific fuel consumption, SFOC, and thermal efficiency of all basic types of main engines fitted to merchant ships. It is clear that in all commercial ship design scenarios, in which the installation of low speed diesel engines is feasible in terms of required engine room volume and weight, the low speed diesel engine will be the preferred type of engine and medium speed diesel engines will follow after. This is evident both from the point of view of lower specific fuel consumption for the low-speed diesel engines (see Fig. 5.1), and the lower price of their fuel; note that the cost for heavy fuel oil—HFO, which is the prime fuel for low-speed diesel engines, was about $360/t in July 2009 and $ 602/t in June 2014 (Rotterdam), whereas that of marine diesel oil—MDO for me-

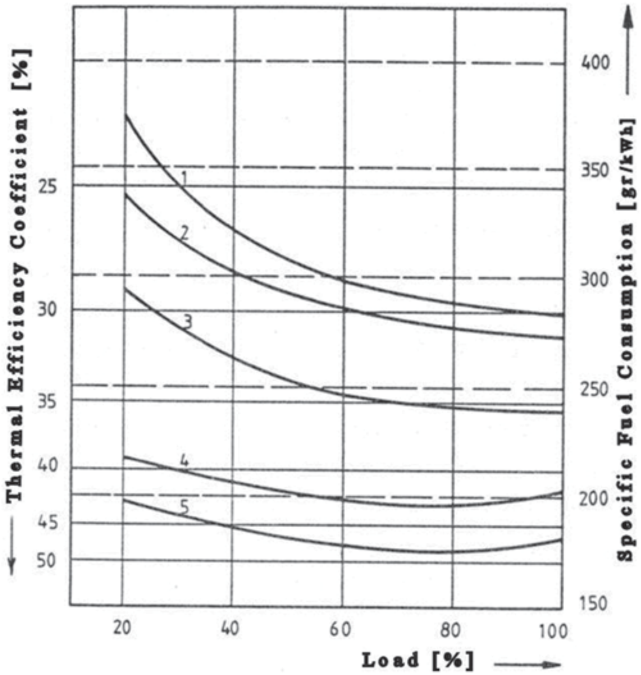

Fig. 5.5 SFOC and thermal efficiency of marine engines: *1* gas turbine, *2* steam turbine, power of 12 MW, *3* steam turbine, power of 30 MW, *4* medium-speed diesel engine, *5* low-speed diesel engine (Schneekluth 1985)

dium speed diesel engines was about \$472/t respectively, \$915/t in June 2014 (very drastic increase of fuel prices, while developments in medium speed diesel engines technology to run with inferior quality fuels are acknowledged).

Comparison of the Fuel Consumption of Different Transportation Means (Schneekluth 1985**)** A comparison of the fuel consumption of alternative (and partly competing) modes of transportation is outlined in the following. The following fuel consumptions refer to 100 km travelled distances with *non-stop and fully loaded* transportation means. Therefore the below benchmark is for an idealized condition, given the fact that neither the various transportation means are equally fully loaded, nor the fluctuations of loadings are the same. The reference fuel is *diesel oil,* except for the cases of airplanes and hovercraft ships, which are considered to run on *aviation kerosene* (gas turbine engine); ocean going ships are considered to run on *heavy fuel oil* (low speed diesel or steam turbine engines) (Tables 5.4, 5.5 and 5.6).

The above tables clearly illustrate the superiority of the ship as the most efficient (in terms of fuel consumption) and more environmentally friendly (in terms of gaseous emissions) transportation mean, particularly for merchandise and bulk cargo. However, it should be borne in mind that the above data do not take into account the transportation speed, which is important for high-value products and passengers, with high '*value of time*'.

Table 5.4 Specific fuel consumption for transport of break bulk cargo

Specific fuel consumption for transport of break bulk cargo	
Ship	0.4 kg/(t 100 km)
Truck	1.1–1.6 kg/(t 100 km)
Train	0.7–1.6 kg/(t 100 km)
Airplane	6–8 kg/(t 100 km) with respect to the airplane's transport capacity (including the weight of fuel)
	11–14 kg/(t 100 km) with respect to the payload for transatlantic flights

Table 5.5 Specific fuel consumption for passenger transportation (vehicles fully loaded, except otherwise mentioned)

Specific fuel consumption for passenger transportation (vehicles fully loaded, except otherwise mentioned)	
Private car, only the driver as passenger	about 8 kg/(person 100 km)
Bus (55 passengers, speed 100 km/h)	0.5 kg/(person 100 km)
German Intercity type train (10 wagons of 60 seats, 160 km/h)	3 kg/(person 100 km)
German D type train (14 wagons of 72 seats, 140 km/h)	1.5 kg/(person 100 km)
Airplane in transatlantic flight (with other load)	17 kg/(person 100 km)
Airplane in European flight (no other load)	3.6–6 kg/(person 100 km)
Hovercraft ship (600 passengers)	5 kg/(person 100 km)
Contemporary cruise ship (500–1,000 passengers)	16–18 kg/(person 100 km)
RoPax ship with passengers on deck (1,500 passengers)	5–6 kg/(person 100 km)
Small river boat, with passenger seats on deck	1.5 kg/(person 100 km)
Large river boat, with passenger seats on deck	0.5 kg/(person 100 km)

Table 5.6 Specific fuel consumption of modern aircraft per seat per kilometre. (Source: Scandinavian Airlines 2012, Traveller's Guide)

Model	Maximum takeoff weight (metric tonnes)	Cruising speed (kilometre per hour)	Range (kilometre)	Fuel consumption (litre per seat per kilometre)
Airbus A340-300	275	875	12,800	0.039
Airbus A330-300	233	875	10,100	0.033
Airbus A321-200	89	840	3,800	0.029
Airbus A319	75.5	840	5,100	0.033
Airbus A320	73.5	840	3,500	0.029
Airbus A737-600	59.9	840	2,400	0.038
Airbus A737-700	69.9	840	4,400	0.032
Airbus A737-800	79	840	4,200	0.028
Airbus A737-400	63	800	3,150	0.034
Airbus A737-500	57	800	4,100	0.039
MD-82	67.8	820	2,400	0.041
CRJ900 NG (next generation)	38	840	2,100	0.039
Boeing 717	53.5	820	2,800	0.037
Dash 8-Q400	29.2	625	1,500	0.034
Dash 8-Q300	19.5	485	1,500	0.034
Dash 8-Q200	16.5	485	1,500	0.038
Dash 8-Q100	15.6	515	1,500	0.038

B. Selection of the Propulsive Installation During the preliminary design stage, the knowledge of the weight and the required space for the main engine, as well as for auxiliary elements of the engine installation, including the fuel weight, is of prime importance. This applies in particular to fast ships (with high requirements on the installed power and substantial quantities of carried fuel), as well as to small ships, due to the increased importance of the above factors on the economic operation of the ship and on the distribution of ship's main weight groups.

The estimation of the machinery weight has been described in previous sections (see Sect. 2.15.6) .

For diesel engines, it is noted that there is an upper limit of maximum delivered power per engine, which has reached the level of about 130,000 bhp for low-speed engines[2]. If the required power exceeds the above limit or there is not available space and weight margin for the installation of a large low speed diesel engine, then the following options need to be explored:

1. Combination of power of more diesel engines through gearbox (applies mainly to medium-speed engines).
2. Multi-propeller propulsion installation with direct drive (one low-speed diesel engine per propeller) or through gearboxes (medium-speed diesels) .
3. Fitting of steam turbine. It is assumed that the selection of steam turbine should be considered as a competitive option in all cases of high power requirement (> 50,000 hp), due to the relatively low weight and simplicity of maintenance and operation.

Nevertheless, the rapid evolution of medium speed diesel engines (independently of the continuous growth of the upper limit of the horsepower of low-speed engines), especially their low weight/space requirements and their continuously decreasing specific fuel consumption, coupled with the high power output per cylinder, have made the medium-speed engines very competitive against all others for all the required power range of contemporary merchant ships (Fig. 5.6).

The use of gas turbines for merchant ships has almost disappeared in practice in the last 30 years, due to the dramatic increase of fuel costs and the high specific fuel consumption of gas turbines; exceptions from this rule are high-speed crafts, demanding very low-weight and limited-space engines and naval ships in general (gas turbines come often as 'boosters' in combined diesel and gas (CODAG) installations) (Figs. 5.7 and 5.8).

C. Specifications in the Shipyard–Owner Contract In general, the technical specifications of the contract between the shipyard and the interested ship owner specify a particular engine installation,[3] as well as ship's speed in the *full load (at design draft) trial condition.*

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[2] MAN B&W, 14K98MC with 108920 bhp at 94(r/min), 14K98MC-C with 108,640 bhp at 104 (r/min), 14K108ME-C with 132,300 bhp at 94 (r/min). Comparable performance have the SULZER engines (e.g. 112RTA 96C).

[3] It is assumed that a study (numerical estimation and verification by model experiments) on the required power to achieve the specified speed has been conducted prior to the contract.

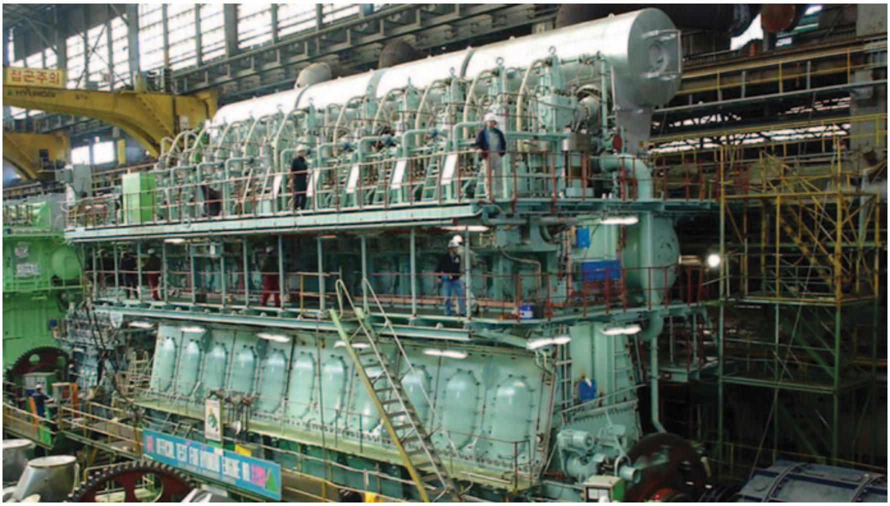


Fig. 5.6 Low-speed diesel engine MAN B&W 12K98MC (2004). 68,647 kW/93,360 PS, 94 rpm maximum. Heavy fuel oil: ISO-F-RMH, maximum viscosity 700 cSt. Consumption: 230 t/day for a 25-knot post-Panamax container ship

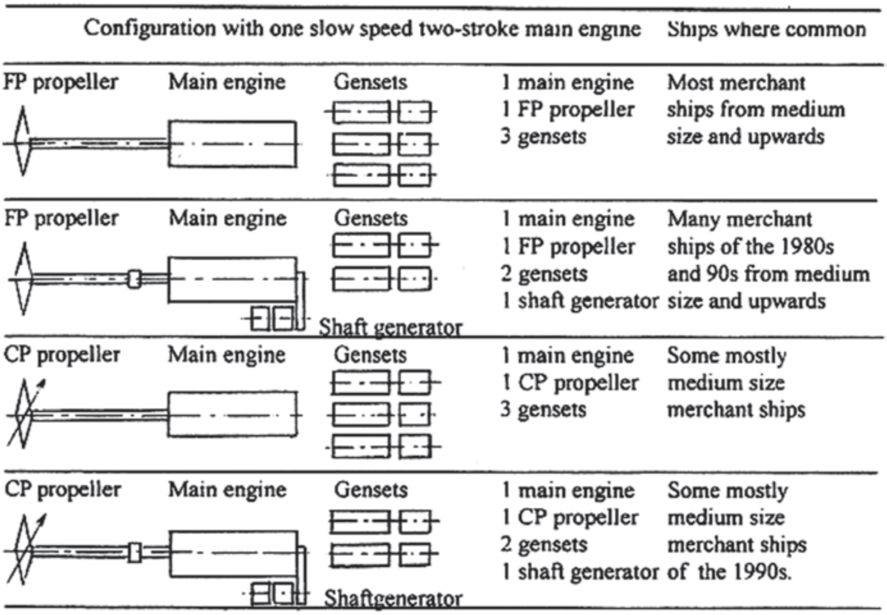


Fig. 5.7 Typical arrangements of low-speed (two-stroke) diesel engines directly connected to either fixed or controllable pitch propeller, including arrangements of generator sets (Dudszus and Danckwardt 1982)

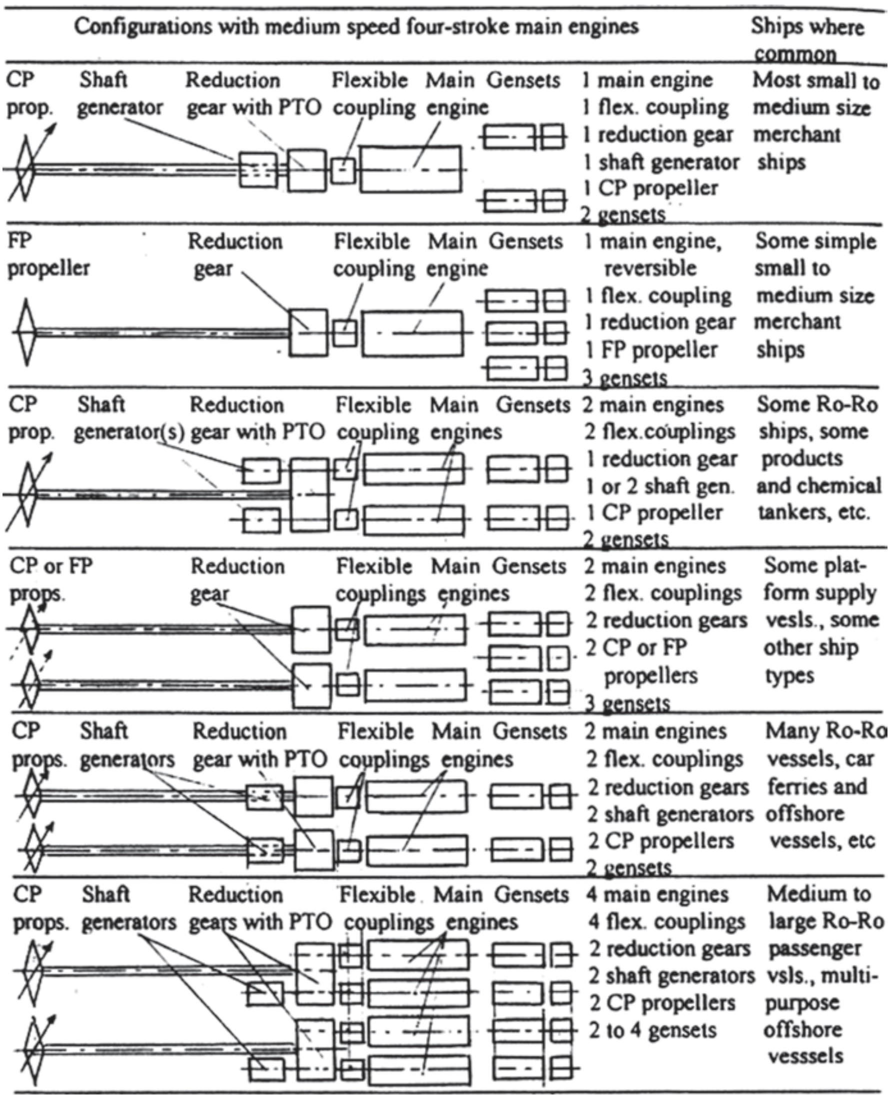


Fig. 5.8 Typical arrangements of medium-speed (four-stroke) diesel engines connected via reduction gear(s) to propeller(s) of fixed or controllable pitch, including arrangements of generator sets (Dudszus and Danckwardt 1982)

The specified/nominal engine power is defined according to the international ISO regulations as the maximum continuous power that the engine can deliver without interruption at corresponding revolutions and under conditions specified by the manufacturer. Interruptions for the necessary maintenance or repairs, which are prescribed by the manufacturer, are not taken into account. The above power is known with the abbreviation MCR (maximum continuous rating). Typical prescribed environmental conditions associated to MCR are (Table 5.7):

Table 5.7 Typical prescribed environmental conditions associated to MCR

Atmospheric pressure:	1000 mbar
Engine room temperature of:	45 °C
Relative humidity of engine room:	60 %
Temperature of feed air and cooling water:	32 °C

The in-service loading and the rating of the delivered power of diesel engine machines, assumed today in the design condition, are usually 75–85 % MCR for the following reasons:

1. The specific fuel consumption is minimal for diesel engines in this region (see Fig. 5.5).
2. The wear of the machine and the repair costs decrease significantly for reduced engine loading of less than nominal 100 %.
3. If a fixed-pitch propeller is used, there is a fundamental problem to deliver the installed full MCR under service conditions in view of the gradual change of the operating conditions of propeller and engine. Specifically, due to the gradual resistance increase (fouling of the hull, rough sea etc.), a change of the relationship: required power $P_B = 100\,\%$ MCR $\Rightarrow 100\,\%\ n_M$ (engine revolutions) is concluded. Thus, the availability of approximately 15–25 % power margin can cover possible *increases of propulsion power demands for maintaining the service speed*.

The above deliberations regarding the availability of power margin are nowadays further enhanced by considerations of *slow steaming* for certain transportation scenarios accounting for the high fuel prices and the competitive market conditions.

For every diesel engine (Fig. 5.9), the manufacturer informs the operator of the regions of engine speed-power for safe operation, both for the continuous operation (CSR: continuous service rating), the maximum continuous operation (MCR: maximum continuous rating) , and the regions of limited/short time of operation (region 2: permitted/allowed short time of operation at reduced loading, region 3: permitted/allowed short time of overcharge of the engine).

Characteristic operational data of different types of diesel engines are given in Papanikolaou and Zaraphonitis (1988) and in known manufacturers' web sites.

D. Selection of Main Engine We consider that the required power of the main engine, which must be delivered and absorbed by ship's propeller for developing a speed *V*, is pre-determined by theoretical predictions and (in practice) validated subsequent model tests, that is, we have the $P_D = f(n)$ curve, where P_D: delivered horse power and n: propeller revolutions. In Fig. 5.5 (Dudszus and Danckwardt 1982) below, the following refers to the various shown curves.

1: trial condition, reduced draft, calm water;
2: ballast condition, relatively calm sea;
3: usual loading condition (70 % DWT), moderate hull fouling, moderate sea state;
4: fully loaded (100 % DWT), significant hull fouling, high seaways;
5: sailing in shallow waters;
6: towing condition, zero speed.

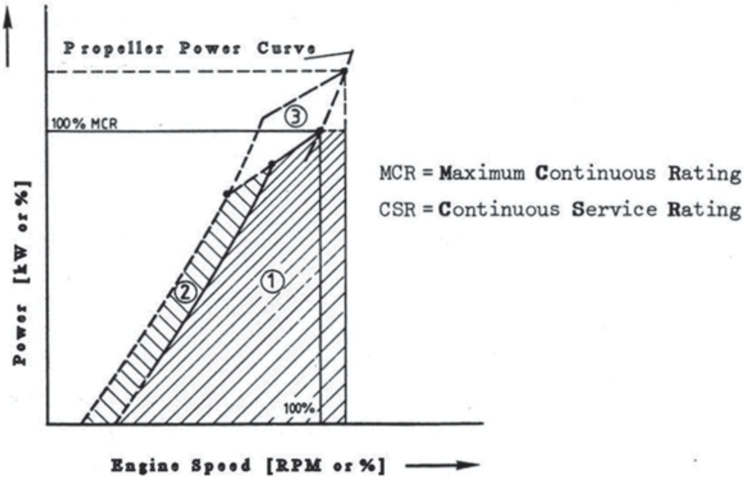


Fig. 5.9 Typical engine speed–power diagram of a ship's diesel engine with typical regions of operation

In addition, for the engine to be selected, the maximum brake power P_B is considered, as given from the data of the manufacturer, where P_B = break horse power and the effective power $P_e = f(n_M)$, where P_e: effective horse power and n_M: corresponding revolutions of the main engine.

The delivered power to propeller $P_D = f(n)$ curves are of parabolic type. In contrast, the corresponding characteristic lines of diesel engine's operation are straight $P_e = f(n_M)$, where P_e is the effective power of main engine, n_M: revolutions of main engine. The marked straight lines $P_e = f(n_M)$ in the previous graph correspond to various ratios of delivered engine's *torque/moment* (M_d) or cylinder *pressure* (p_e), recalling the known relationship (Dudszus and Danckwardt 1982):

$$M_d = \frac{1}{2\pi}\frac{p_e}{n_M} = \frac{V_M \cdot p_e}{2\pi \cdot i} \tag{5.4}$$

where

M_d Delivered engine torque/moment (kilonewton-metre)
P_e Effective power (kilowatt)
n_M engine revolutions (1/s)
p_e mean effective cylinder pressure (megapascal)1 MPa = 106 N/m^2
V_M volume of engine's cylinders (cubic metre)
i 1: two-stroke, 2: four-stroke diesel engine

From the above relationship it is concluded for the effective power P_e:

$$P_e = V_M \cdot p_e \cdot i^{-1} \cdot n_M \tag{5.5}$$

namely, for constant cylinder pressure p_e, of cubic capacity V_M, or for constant torque/moment M_d, there is a linear relationship:

$$P_e \propto n_M.$$

From the difference in the character of the curves $P_D=f(n)$ and $P_e=f(n_M)$, it is evident that an excess of the generated effective moment of the engine, even for reduced revolutions n_M in relation to the nominal (100 %: nominal speed), may result.

Thus the manufacturer specifies the following operational regions and characteristic points of the engine:

a. Maximum continuous power, maximum brake power (P_B), nominal power P_e

$$P_e = \text{MCR}$$

This power corresponds to 100 % revolutions of the engine (nominal speed). It is recommended to avoid exceeding the MCR, in all cases, thus the operating points under service conditions are in the region of about 75–85 % MCR.

b. The operating region I corresponds to the usual, continuous service (CSR). This region is bounded by the lines of 100 % P_B and 100 % M_d (or p_e).
c. The operating range II is only for limited duration (torque limit, operating range temporary admissible), e.g. during the acceleration or manoeuvres.

Notes (Fig. 5.10, Dudszus and Danckwardt 1982)

1. The vertical scale on the right side of the diagram refers to the ship's speed V in conditions 1 and 3, in relation to the trial speed ($V_T \equiv V_{1,0}$ for 100 % MCR).
2. The speeds achievable in the conditions 1 (trial), 3 (service) for engine output 100, 75, 50 and 25 % MCR (see curves 1 and 3) are given on the same scale.
3. The indices (1,0) mean:

$V_{1,0}$: trial speed
$n_{1,0}$: propeller revolution for $V_{1,0}$ or nominal engine revolutions
$P_{D1,0}$: delivered power to the propeller for $V_{1,0}$.

5.2 Selection of Propeller

A. General Issues

A.1. Fixed Pitch Propellers

History The fundamental idea of the function of a *screw/propeller* (and its companion the *impeller*) as a means to lift water for irrigation or thrust generated by fluids is due to the Great Greek mathematician, physicist, astronomer, engineer and innovator *Archimedes* (287–212 BC), when he introduced his screw pump (Fig. 5.11).

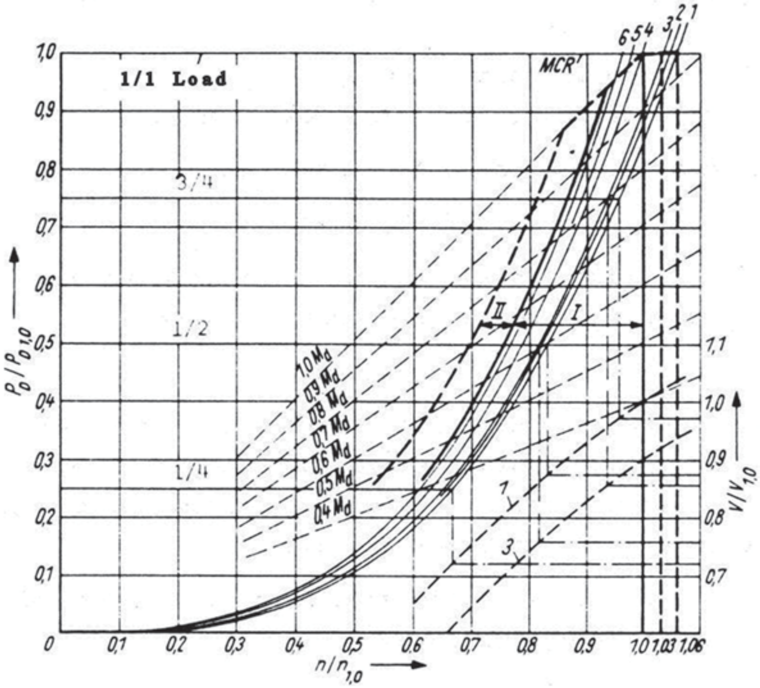
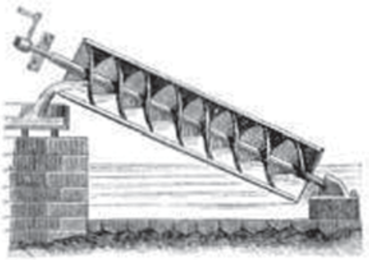


Fig. 5.10 Typical curves of propeller and diesel engine powering vs. propeller/engine revolutions for a cargo ship according to Dudszus and Danckwardt (1982); Region I: continuous operation without restrictions; Region II: short operation within time limits set by the manufacturer

Fig. 5.11 Archimedes' screw pump



Long after, the eminent scientist Daniel Bernoulli[4] (1730) proposed the propeller as propulsive means of floating vehicles. Almost 100 years later, *R. Wilson, F. Sauvage, J. Ericsson and F. P. Smith*[5] refined D. Bernoulli's proposal, so that a few

[4] Daniel Bernoulli (1700-1782): Eminent Swiss mathematician and physicist, with pioneering contributions to fluid mechanics (conservation of energy, Bernoulli equation) and to the theory of probability and statistics (St Petersburg Paradox; risk theory); in his work 'Hydrodynamica' (published in 1738) he laid the foundations of the theory of watermills, windmills, water pumps and water propellers.

[5] *J. Ericsson and F. P. Smith* filed in parallel controversial patents for the use of propeller as propulsive means of ships (1836); *Francis Pettit Smith* discovered the way of building screw propellers

Fig. 5.12 SS Great Eastern (launched 1858), propelled by two side paddle wheels plus one screw propeller

decades later the propeller replaced the paddle wheels as main propulsive means of ships. The first large ship to be fitted with a screw propeller was the famous *SS Great Eastern* of Isambard Kingdom Brunel. She was by far the largest ship ever built at the time of her launch, in 1858; she had a capacity of 4,000 passengers and could sail around the world without refuelling at a speed of 14 knots (Fig. 5.12).

a. **Main Features** (Comparison: propulsion with propeller against earlier used propulsion systems, e.g. side paddle wheels) :

- High efficiency
- Easy adaptation to alternative hull form designs and ship's operation
- Weak effect of seaway on its performance
- High number of propeller revolutions
- Protected location: less exposed, below the free surface at the stern
- Small disturbance on the general arrangement of the ship
- Small weight
- Possibility of receiving large delivered propulsion power; today, up to about 75,000 hp[6] per propeller shaft.

b. **Number of Blades**

Two to six (seven) blades per propeller
Two: fast small boats, outboard engines
Three: multi-propeller ships, fast passenger ships, naval ships
Four: ordinary cargo ships
Five: sometimes for high-powered cargo ships, reduce vibration level
Six: rare, occasionally for high powered large ships and old transatlantic ocean liners, e.g. the former 'Queen Elisabeth II'
Six to seven: large naval submarines (nuclear-powered).

of the type we know them today accidently. Up to that time, propellers were literally screws, of considerable length. But during the testing of a boat, the screw broke, leaving a fragment shaped much like a modern boat propeller. The boat moved faster with the broken propeller; this event may have led us to the ship propellers of today!

[6] MEGA Containership Project: estimated propulsive power about 100,000 hp, diameter of propeller (if single screw) up to approximately 12 m (according to estimations of Lloyd's Register).

Fig. 5.13 The Rolls Royce Kamewa adjustable pitch propellers

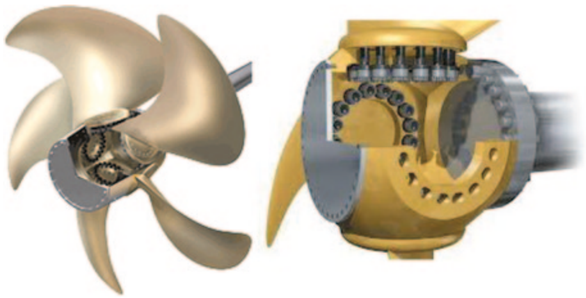


Fig. 5.14 Large, high powered, fixed-pitch propeller for post-Panamax container ship (2004). Diameter 9.10 m; six blades, total weight 102 t

c. Construction

Earlier times (until around the World War II): Individual blades bolted to the propeller hub.

Advantage: Easy casting of blades, easy repair; slight change of the pitch during docking possible.

Most recent developments: The Rolls Royce Kamewa adjustable pitch propellers (ABP) (Fig. 5.13); the concept is based on a hollow hub with blades bolted to it from the inside. In comparison to conventional monoblock fixed-pitch propellers (Figs. 5.14 and 5.15), the ABP has higher-quality blade machining and reduced overall weight, which results in easier shipment, handling and mounting. The slotted holes on the hub allow the blade pitch angle to be conveniently adjusted at commissioning, or in service to compensate for long-term

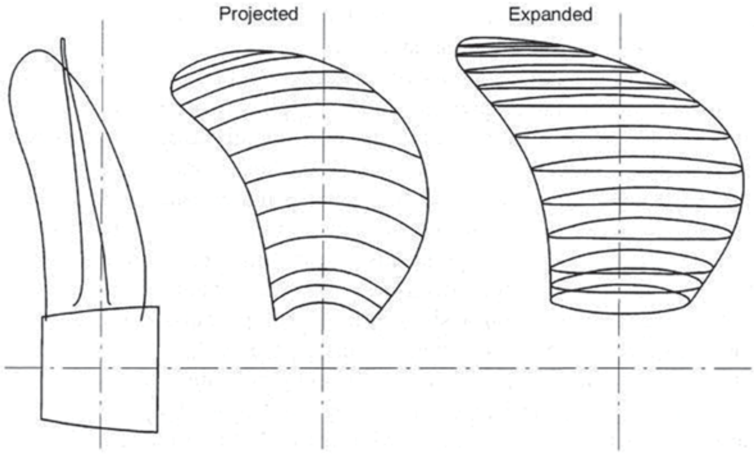


Fig. 5.15 Fixed pitch propeller geometry (Friis et al. 2002)

variations in hull resistance. Individual blades can be replaced without dry-docking, and only spare blades have to be stocked rather than a complete monoblock propeller.

Nowadays: Mostly casted, fixed-pitch monoblock propeller.

Advantage: Small hub: higher efficiency, better absorption of bending moments of the blades at the base of the hub

Construction material: Mainly manganese bronze and other copper-tin alloys.

A.2. CPP (Controllable Pitch Propellers)

a. **Main features:** Direct connection of engine's operational profile to the propeller for maximum absorption of generated horsepower, without changes in the engine and propeller's revolutions, by adaptation of the pitch of the propeller to the various operating conditions and thrust demand of the ship. It also eliminates the need for a reversing gear and allows for more rapid change to thrust, as the revolutions are constant.

Characteristics of Engines:

- **Reversing Capability**

Diesel engines: Easily reversing, but causing high thermal loading due to the low temperature of the supplied air at the startup

Steam turbine: Reversing power about 40 % of that ahead, development of thermal stresses at the bearings of the blades

Gas turbines: Irreversible

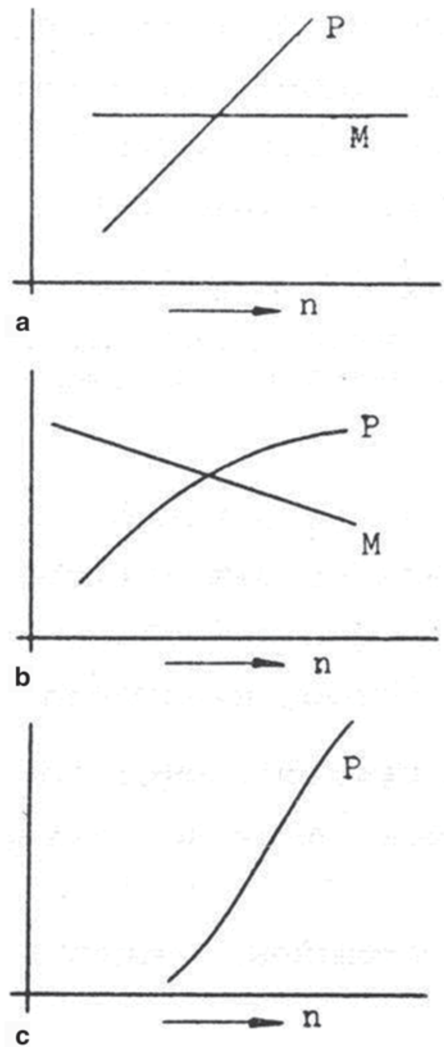


Fig. 5.16 Relationship of torque vs. engine revolutions for diesel (**a**), steam turbine (**b**) and gas turbine (**c**)

- **Relationship of Torque vs. Engine Revolutions**

  Diesel engines: For constant fuel rating we obtain constant torque and a linear relationship between the effective power and the number of engine revolutions (see Fig. 5.16a).

  Steam turbines: Decreasing torque with increasing number of engine revolutions, gradual increase of effective power (see Fig. 5.16b).

  Gas turbines: Rapid increase of effective power with the increase of engine's revolutions (see Fig. 5.16c).

- **Minimum Revolutions**

  Diesel engines: About 30 % of the nominal number of revolutions (nominal speed)

  Turbines: Very low limit compared to the nominal speed

B. Application of CPPs

Large Changes of Thrust Demand CPP is commonly applied to fishing vessels or tug boats, because of their two totally different operating conditions: *free cruising at high speed* to the service area (e.g. fishing area or work/assistance area for tugs) and *towing condition* (e.g. trawling of fish net or towing of other vessels by tugs) *at low speed* and high thrust demand. In the case of applying *fixed-pitch* propeller in towing condition, the engine revolutions may reduce by about 70 % and correspondingly the delivered power of the main (diesel) engine drops. Also, due to the resulting high thrust rating (high thrust T at low speed), the efficiency of the propeller is significantly reduced and may drop down to 30 %; thus, the total efficiency of the propulsive system (machine–propeller) is very low. With the application of CPP, the propeller pitch can be reduced so as to keep a high number of propeller revolutions and consequently of the speed of the engine, with high efficiencies on both ends.

Comparable heterogeneous conditions do not present for normal merchant ships in their operation, and therefore, it is not recommended to use CPPs in these cases, because of the involved additional installation costs and some other drawbacks. Exceptions to this rule are safety critical vessels, like passenger ships, where CPPs contribute to excellent steering/manoeuvring capability (berth manoeuvring in limited waters), thus typically they are applied as twin CPPs to all contemporary passenger ships.

Variable Engine and Propeller Loading of Naval Ships Particularly for naval ships, two characteristic operating conditions prevail:

Cruising at marsh speed: Requirement for low fuel consumption and operating cost, which is achieved by combined diesel and/or gas or steam turbine machines

Cruising at top speed: Requirement for the availability of additional, relatively light engines (*boosters*), which can boost the ship to top speed; e.g. gas turbines and CODAG systems.

If the propulsion plant considers the synchronous or individual use of different engines/machines for the marsh and top speed, the use of CPPs is practically imperative.

C. Efficiency of CPPs As the *radial* distribution of the pitch of the propeller blades is optimal for *only one* blade position, it is clear that for other positions of the blades, except for the optimal one, the efficiency decreases. In addition, the relatively large hub of CPPs *negatively* affects the propeller efficiency.

D. Operating Modes with CPPs CPPs can operate with constant speed/revolutions, which is anyway necessary if the propulsion plant is directly connected to electric generators. In the case of autonomous electric generator sets, it is attempted to achieve optimum propeller performance with the synchronous change of the propeller pitch *and* revolutions (through one single control lever). In the low revolution

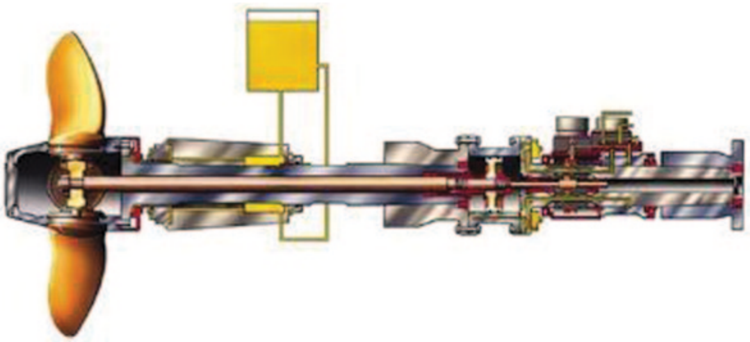


Fig. 5.17 Controllable pitch propeller

range, only the pitch adjustment is possible. Modern diesel–electric propulsion plants (mainly on passenger ships) allow the operation of the propulsion and electrical systems with best performance for both the engines and for the propulsive means.

Attention To keep *zero* thrust, with the engine turning, the blades of the CPP must turn almost perpendicular to the axis of the propeller; thus their projected disk area overlaps the rudder of the ship, preventing the water flow to the rudder; thus, due to the lack of a lifting force on the rudder, the manoeuvrability of the ship greatly reduces (Fig. 5.17).

B. Number of Propellers
B.1. Selection Criteria

a. **Weight, space requirements, cost:** A multiple-propeller ship is economically inferior to a single-propeller ship for the following reasons:

1. More bearings for engine, shafts etc.
2. Larger required engine room, if engine placed abaft
3. Higher space requirements in general (two propeller shaft tunnels)
4. More auxiliary machinery, piping etc.
5. More personnel, maintenance/repair effort
6. Higher weight
7. Higher acquisition/installation cost

b. **Number of engines:** The limited power of a single diesel engine, for instance, low-speed engines have today an upper limit of about 130,000 hp, but also the limiting value of maximum absorbed power by a single propeller, which is at about the same level, requires the installation of more than one propellers on specific types (and sizes) of ships with generally high horsepower requirements (e.g. large and fast container ships, large and fast RoPax ships, ultra large tankers and naval ships) .

For very high horsepower requirements on large ships, beyond a set of more than one low-speed diesel engines, or of medium speed engines, it is advisable to consider the use of steam turbines, with practically unlimited maximum delivered power.

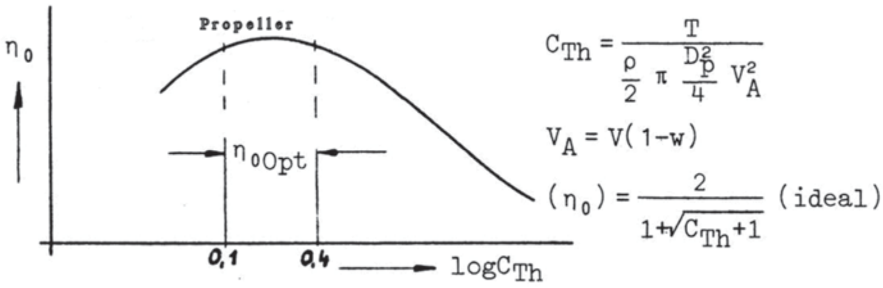


Fig. 5.18 Qualitative relationship of thrust coefficient C_{Th} and propeller efficiency

c. **Propulsion redundancy:** Multi-propeller and multi-engine ships have the propulsion ability even in case of failure of one unit. Especially for RoPax and passenger ships in general, this is an important feature ensuring safe operation (see requirements of Safety of Life at Sea (SOLAS), 'safe return to port' provisions) .
d. **Steering:** In case of rudder failure, a twin-screw ship is able to perform limited manoeuvring by setting different revolutions for the two propellers. Also, opposite (contra-)rotating propellers can turn the ship in restricted waters (e.g. berthing at ports etc.). RoPax ships engaged in frequent manoeuvres at limited berthing ports are required according to SOLAS to dispose two independent propellers driven by two independent main engines.
e. **Exploitation of wake losses:** As known, the frictional part of a ship's wake decreases significantly in the transverse direction, as we depart from the ship's centreplane. Thus, ships with at least one middle propeller have higher hull efficiency factor η_H, than twin-screw ships.
f. **Propeller efficiency:** The propeller efficiency depends mainly on the thrust loading coefficient C_{Th}. Generally, in the common range of propeller operation, the propeller efficiency η_0 decreases significantly with the increase of C_{Th} and of propeller loading (see Fig. 5.18).

Thus, in case of high required thrust levels, particularly for low-speed design scenarios, i.e. high C_{Th} coefficients, the installation of two propellers is preferable, because this way the resulting C_{Th} is reduced and the efficiency η_0 increases. Thereby, it is assumed that the maximum allowable ship's draft does not allow the installation of a single propeller of large diameter (maximum diameter: approximately 65–70 % of the ship's draft). The resulting relatively high efficiency η_0 of the multi-propeller ship compensates the aforementioned reduction of hull efficiency η_H.

Another way of reducing C_{Th} and thus increasing the propeller efficiency in high-thrust/low-speed conditions is the fitting of *Kort* type nozzles, leading to *ducted-propellers,* in which the onset flow speed to the propeller is increased and the propeller loading and C_{Th} decrease; it is applied frequently to tug boats and fishing vessels (Fig. 5.19).

g. **Conclusion:** In general, the number of propulsive means should be kept at minimum. Taking into account the specific design and operational conditions of the study ship (requirements of space arrangement, thrust and speed conditions and

Fig. 5.19 Ludwig Kort's nozzle (invented 1934) and ducted propeller

magnitude of propulsive power) multi-propeller installations may be concluded; in that case, the propulsion installation must be carefully designed (for optimal water onset flow) to compensate for the additional costs for the propeller fitting and the operation of the ship.

B.2. Typical Number of Propellers

Cargo ships: commonly one, occasionally two for fast ships
Bulk carriers: always one, rarely two (large ships)
Tankers: usually one, large very large crude carrier (VLCC) and ultra-large crude carrier (ULCC): two
Short-sea cargo ships: always one
Container ships: one to two (depends on size), in the past up to three
Reefers: mostly one
Past transatlantic ocean liners: mostly four, occasionally two
RoPax: commonly two
Tugs: mostly one, occasionally two
Icebreakers: two, occasionally four
Naval ships: mostly two, occasionally three and four.

C. Arrangement of Propellers

C.1. Overview

a. Astern or Bow Propellers

The placement of the propeller at the stern of the ship is preferable for the following reasons:

1. The pitch and absolute motions of the stern of the ship are smaller than the corresponding motions of the bow; thus likely propeller emerging in seaways is reduced.
2. The protection of the propeller against damage is much higher at the stern.

3. Though the open water efficiency η_O of a bow propeller may be higher in calm water conditions, the overall propulsive efficiency of a stern propeller is higher, because through the stern propeller it is possible to regain part of the lost energy corresponding to the frictional part of the wake of the ship. This is reflected in the relatively high propulsive coefficient η_D.
4. The ship's general arrangement is less disturbed.

The application of bow propellers is sometimes seen on double-ended, small car ferries, with symmetrical bow and stern propulsive and navigational arrangements, and also on some icebreakers.

b. **Propeller Diameter**
As the diameter of the propeller increases, the thrust coefficient C_{Th} decreases rapidly with the square of the diameter, and consequently the efficiency increases. However, simultaneously, the peripheral speeds at the tips of the propeller blades also increase, in other words, the local hydrodynamic pressure decreases, which increase the risk of cavitation. In general and in the absence of other constraints, the propeller diameter is selected in the range of 65–70 % of ship's design draft.

C.2. Single-Screw Ships

a. **Typical diameter sizes:**

Cargo ships 5.0–6.5 m
Reefer ships 4.8–5.3 m
Tankers up to about 10 m
Mega-container ships up to about 12 m (projected)

b. **Propeller position as to the waterline:** The partially (even periodically) emerging propeller induces strong fluctuations of generated thrust and vibrations, due to the trapping of air bubbles close to the propeller blades. It is considered that in the loaded condition, the propeller shaft/axis must be immersed by approximately *one diameter* below the waterline. In ballast condition of ordinary cargo ships, the propeller may emerge by one-seventh to one-third of its diameter. The negative effects of possible propeller emergence (*propeller racing*), which occurs rarely (mainly in ballast condition or in extremely rough seaways), are compensated by the good performance of a large diameter propeller in the full load, design condition. For tankers, due to their frequent cruising at ballast draft (half of their voyages), it is necessary to ensure that the propeller is fully immersed in the ballast condition (see regulations of International Convention for the Prevention of Pollution from ships (IMO MARPOL 73/78 2013)). Also, the optimization of their operational trim in ballast condition is a matter of prime interest to operators, in view of possible reductions of fuel cost expenses.

c. **Clearances between propeller and stern hull:** During the rotation of the propeller and in particular as the propeller blade tips approach the rudder and the stern hull, impulsive loading phenomena occur, which are expressed as fluctuations of propeller's thrust and torque, vibrations and noises, which are transmitted through the propeller shaft to the engine as well as to ship's hull in front of

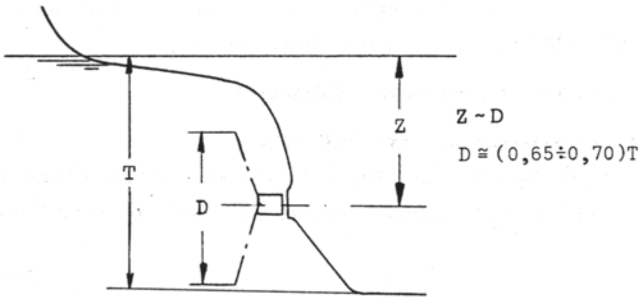


Fig. 5.20 Propeller diameter and magnitude of propeller immersion of single-screw ships

the propeller. Simultaneously, having the propeller astern (*in the wake*) of the hull causes fluctuations of the generated thrust, because of the uneven water onset flow to the propeller.

The minimum clearances between propeller and the neighbouring fittings and the hull of the ship are specified by classification societies and have been already commented on in Sect 3.5. It should be pointed out that since the early 1950s, with the introduction of *Mariner* class general cargo ships, the 'hanging/suspended' rudder without rudder heel has prevailed. Thereby, the induced oscillations are reduced and larger propeller diameters could be fitted with high efficiency (Fig. 5.20).

C.3. Multi-Screw Ships

a. **Rotation direction:** Side propellers always rotate symmetrically, but contra-rotating; generally, looking from astern, from top to down and from inside to outside.
b. **Arrangement and diameter:** Except for the fast naval ships, where the propellers can rotate partially below the keel line, it is recommended for the propeller tips of multi-screw ships to turn a little above the baseline.

   Typical values of propeller diameters D_P and ratios to ship drafts T of multi-screw ships (Table 5.8):
c. **Longitudinal position:** The longitudinal position of the side propellers should be located as sternward as possible, despite the associated extension of the propeller shaft. The internal and external propellers of four-screw ships (e.g. old, fast transatlantic passenger ships) are placed lengthwise shifted. Also, it must be considered that the projections of the disk areas of three-screw or four-screw ships do not overlap in the cross view (see Fig. 5.21and Fig. 5.38).

Table 5.8 Typical values of propeller diameters D_P and ratios to ship drafts T of multi-screw ships

	D_P (m)	D_P/T
Fast (historic) ocean liners:	4.9–6.0	0.45–0.60
Large passenger ships:	4.8–5.6	0.58–0.67
Modern RoPax:	2.4–3.8	0.56–0.73[a]

[a]Modern fast RoPax ships may be fitted with twin propellers of very large diameter, when applying *tunnel-type* sections at the stern

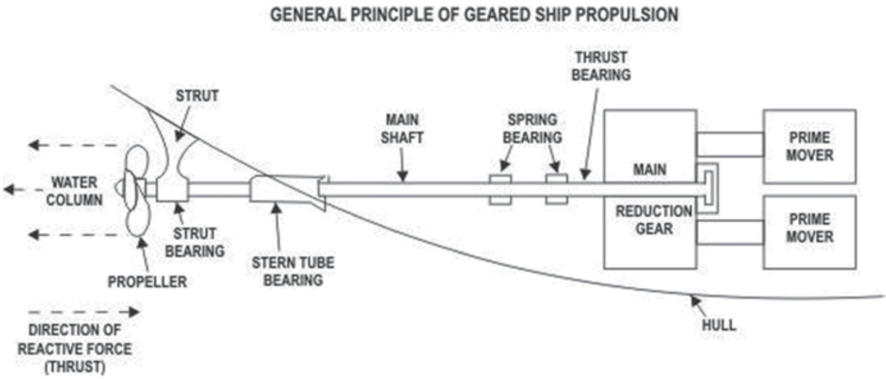


Fig. 5.21 General principle of geared ship propulsion (Lewis 1988)

d. **Position of propeller end shaft and stern tube bearing**: The simplest fitting solution is to install the propeller end shaft parallel to both the basic reference plane (keel) and the ship's centreplane. Differentiations, namely diverging axes as to the aforementioned planes, are often the result of specific arrangements/constraints of the propellers or the machinery.

Regarding the location of the stern tube bearing, there are two conflicting criteria: The bearing should be easily accessible to simplify the installation and maintenance of the propulsion system; however, the length of the end shaft should not exceed an upper limit, which may reach quite large values for large/slender ships.

e. **Exits of propeller end-shafts**

Alternative Fitting Solutions:

a. Long shaft bossing
b. Short shaft bossing and additional support (bracket)

Long shaft bossing is preferred on relatively full hulls (tankers), while on the contrary the fitting method (b) is usually applied to fine-slender ships.

The reasoning for these preferences is as follows: for full-type ships the length of the shaft bossing is limited in practice and the additional fitting of individual brackets is not required for reducing the bossing's wetted surface. In contrast, for fine-slender hulls the bossing would have been enlarged, leading to an increase of ship's viscous resistance, also in view of possible flow separation. Thus, the short bossing with shaft brackets in between is recommended for fine-slender hulls, which reduces the wetted surface and proves to be the more efficient solution, assuming that the cross-section and location of the brackets are properly designed by the conduct of model tests (and/or computational fluid dynamics (CFD) calculations).

Contemporary developments in the hull form optimization of the stern of full type, twin hull ships (e.g. large tankers or bulkcarriers) consider the design of *twin-skeg* stern arrangements, with excellent propulsive efficiency results, if properly optimized. (Fig. 5.22)

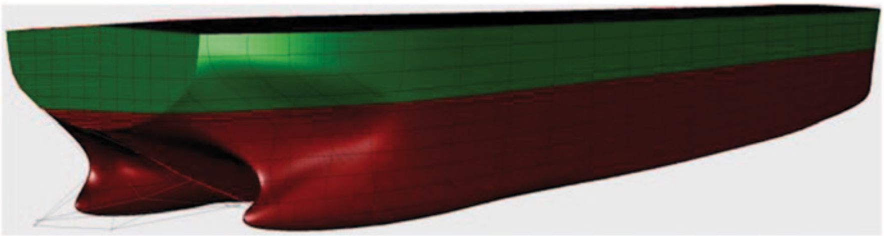


Fig. 5.22 Optimized twin skeg arrangement of innovative tanker (Nikolopoulos, NTUA-SDL 2012)

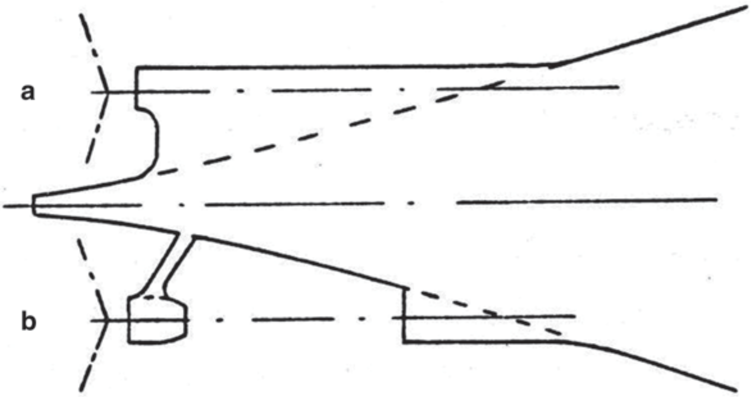


Fig. 5.23 Configuration of shaft bossing for multi-screw ships

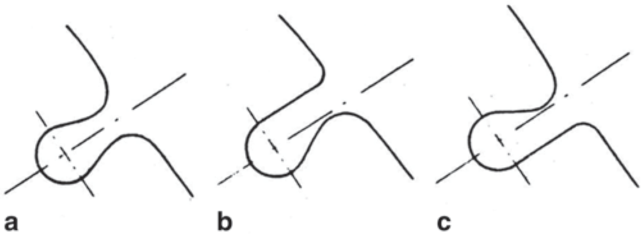


Fig. 5.24 Possible cross sections of shaft bossing for multi screw ships

Configuration of Bossings and Shaft Brackets Common configurations and forms of the cross section of shaft bossings are shown in Figs. 5.23 and 5.24. The symmetric form (a) is considered as the standard solution. Practical implementation are shown in Fig. 5.25.

The transverse symmetry axis of the bossing should be approximately perpendicular to the adjacent hull sections.

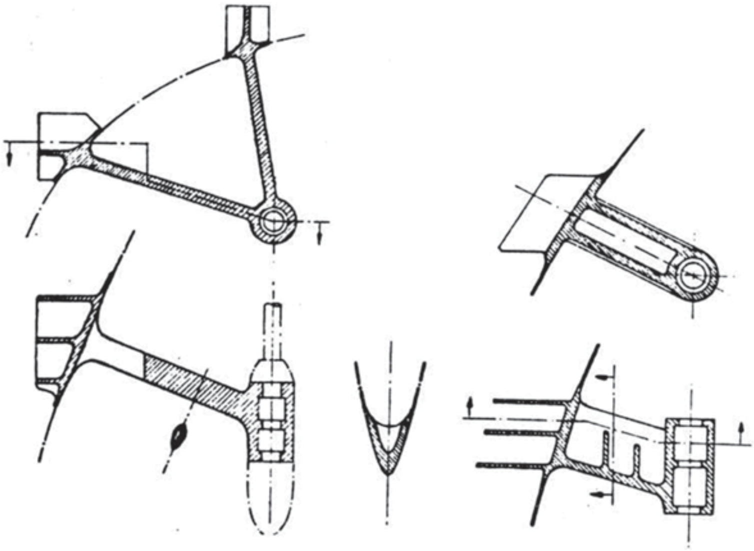


Fig. 5.25 Possible constructional solutions of shaft brackets of multi-screw ships

In the optimization of the arrangements of the stern sections and shaft axes/brackets of multi-screw ships, a smooth water flow *against* the direction of *propeller rotation* (*pre-whirling*) should be targeted. In this way, an exploitation of the induced angular momentum is aimed, so that the energy/resistance losses of the ship are mitigated.

Examples In earlier years of shipbuilding history, the long bossings with complete coverage of the shafts were preferred (see ocean liners 'Bremen' (1929), 'Queen Mary' (1936)). However, in recent times the short bossing with individual brackets and free axes prevail (see naval ships, contemporary passenger ships, container ships etc.). Most recent developments with the fitting of optimized twin skeg arrangements (practically leading back to longer bossings and eventually degenerating to stern bulbs) are notable.

D. Determination of Optimal Propeller The determination of the characteristics of propellers with optimum performance for a given ship and speed (assuming the ship's resistance is known), is described in specialized literature (e.g. Lewis 1988, Politis and Lambrinidis 1993). In the lecture notes (Papanikolaou 2009, Vol.2) the author outlines the procedure for determining an optimal propeller of the Wageningen, B-series, assuming the propeller diameter pre-specified at maximum size ($D_P \approx 0.65$–$0.70\ T$).

E. Contemporary Propulsive Means The explosive development of fast passenger ships and of propulsion technology in general over the past three decades has led to the introduction of a series of unconventional propulsion means with applications to various types of ships (in addition to passenger ships), such as (see, also, subsequent photo series, Figs. 5.26, 5.27 and 5.28) :

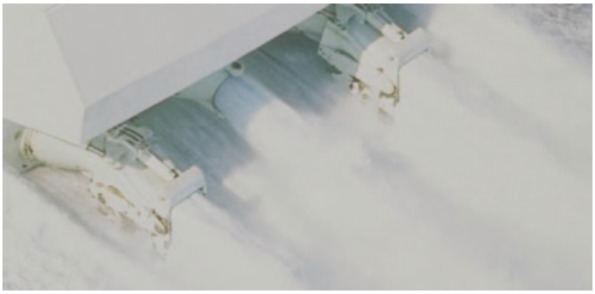


Fig. 5.26 Installation of waterjets on high-speed craft

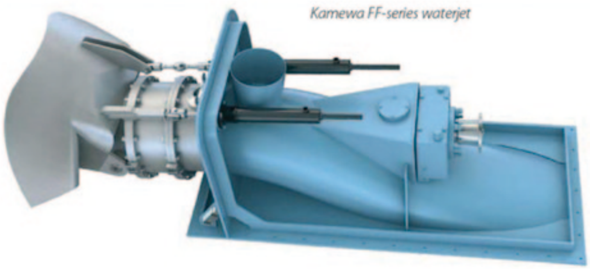


Fig. 5.27 Installation of waterjets on high-speed craft. (concept originally proposed by the Italian inventor Secondo Campini in 1931; the first to apply it commercially was the New Zealand inventor Sir William Hamilton in 1954)

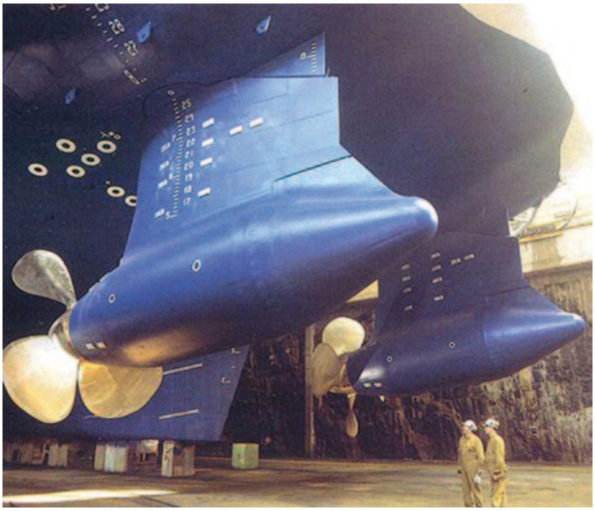


Fig. 5.28 Installation of azimuthal podded drives. The azimuth thruster using a Z-drive transmission was invented in 1950 by the German Joseph Becker, the founder of *Schottel* company. First applications came in the 1960s under the *Schottel* brand name and referred to as Rudder propeller ever since. Later, subsidiaries of ABB, also based in Finland, developed the *Azipod thruster*, with the electrically driven motor located in the pod itself. This kind of propulsion was actually first patented in 1955 by Pleuger, also from Germany.

1. Waterjets
2. Azimuthal podded drives
3. Voith–Schneider propeller system
4. Contra-rotating propellers on the same axis (co-axial)
5. Vane wheel system (patent of late Prof. Otto Grim[7], Hamburg)

The above listed propulsion means (1–3) provide the ship, besides high-efficiency propulsion, significantly improved manoeuvring capabilities, so that there is no need to install a rudder for ship's safe operation. A comprehensive source of information on history and contemporary marine propellers and propulsion may be found in Carlton (2007; Figs. 5.29, 5.30, 5.31 and 5.32).

Fig. 5.29 Installation of Voith–Schneider propulsion system. The Voith–Schneider propeller was originally a design for a hydroelectric turbine. Its Austrian inventor, Ernst Schneider, worked with Voith's subsidiary in St. Pölten to further develop this concept to practical applications. It was found that Schneider's concept worked well as a pump, but, also, by changing the orientation of the vertical blades, it could function as an efficient propeller (first ship prototype in 1928)

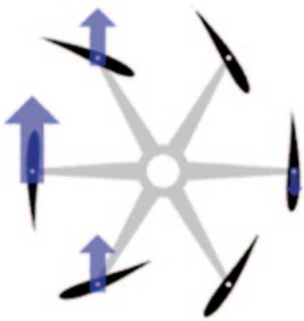


Fig. 5.30 Operational principle of Voith–Schneider Propeller

[7] Grim Otto (1911–1994): Eminent Austrian professor of ship hydrodynamics and director of the Hamburg Ship Model Basin after WWII, with pioneering contributions to ship hydrodynamics, seakeeping, ship vibrations and ship propulsion; less known was his unique expertise and contributions to the structural design of submarines of the German navy during WWII.

Fig. 5.31 Installation of co-axial contra-rotating propellers (CRP). The history of CRPs goes back to 1836, when a patent was filed by *Ericsson* applying it to a 45-foot ship. In 1909 and 1939, the Italian and US navies applied CRP to small steam powered ships. Since then, CRP has been widely used for torpedo propulsion, for small vessels and of course for prop aircraft propulsion; however, mechanical difficulties in producing reliable CR shafting, for large propulsive power have hindered in the past wide application to large merchant ships.

Fig. 5.32 Installation of O. Grim's vane wheel. The *vane wheel* is a second propeller *downstream* of the main propeller, which runs *freely without torque on the shaft*. The *inner part* of the vane wheel, *the impeller or turbine part*, has a pitch such that the vane wheel is driven by the wake of the main propeller. The *outer part* of the blades of the vane wheel, the *propeller part,* has a different pitch, which causes the vane wheel to generate thrust of the main propeller. The concept was originally proposed and patented by the late Prof. Otto Grim (Hamburg) in the 1960s; it is also known as *Grimsche Leitrad*. Vane wheels are subjected to strong fluctuations in loading and problems with the strength of the blades have been encountered frequently, which has led to limited applications in recent years, despite significant powering/fuel savings, when smoothly working.

5.3 Selection of Rudder

A. Overview A rudder, with area A_R gives the ship the turning/evolution moment (Figs. 5.33, 5.34, 5.35 and 5.36):

$$M_{ev} = C_{Ru} \cdot \frac{\rho}{2} V_{Ru}^2 \cdot A_R \cdot a = F_R \cdot a \tag{5.6}$$

where

V_{Ru} Onset flow velocity to the rudder,
V_{Ru} $< V_S$, for rudder outside the propeller flow
$> V_S$, for rudder abaft/within propeller flow
C_{Ru} Rudder lift force coefficient dependent on the rudder foil form, the rudder profile (relationship to aspect ratio) and the incident angle of the water flow
A_R Projected rudder area in the lateral plane
a lever arm of application of the induced rudder force F_R with respect to the centre of mass of the ship,
ρ density of water

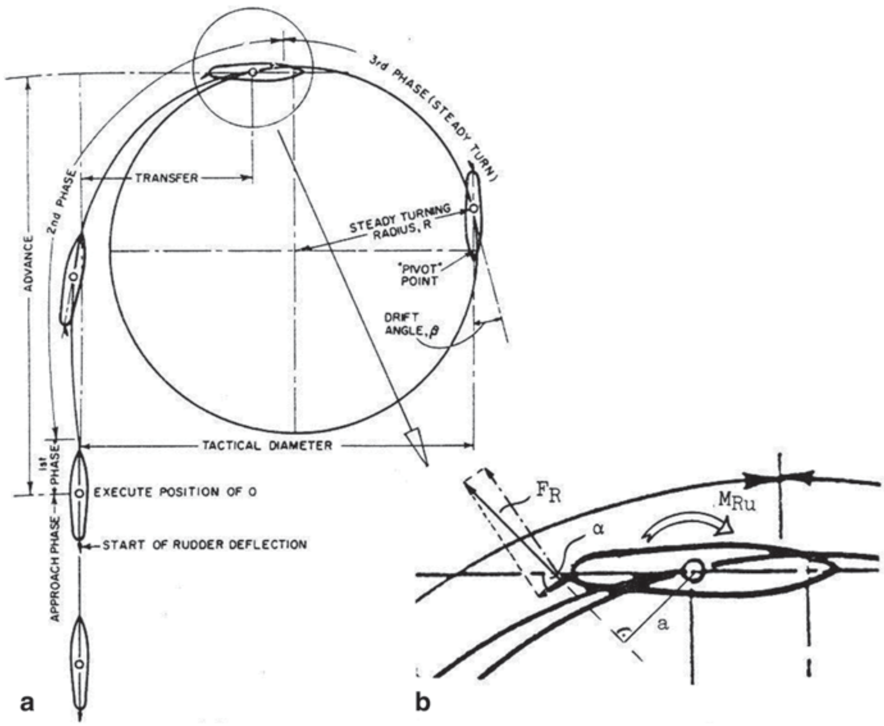


Fig. 5.33 **a** Turning course of a ship (Lewis 1988). **b** Effect of rudder, evolution moment M_{Ru}

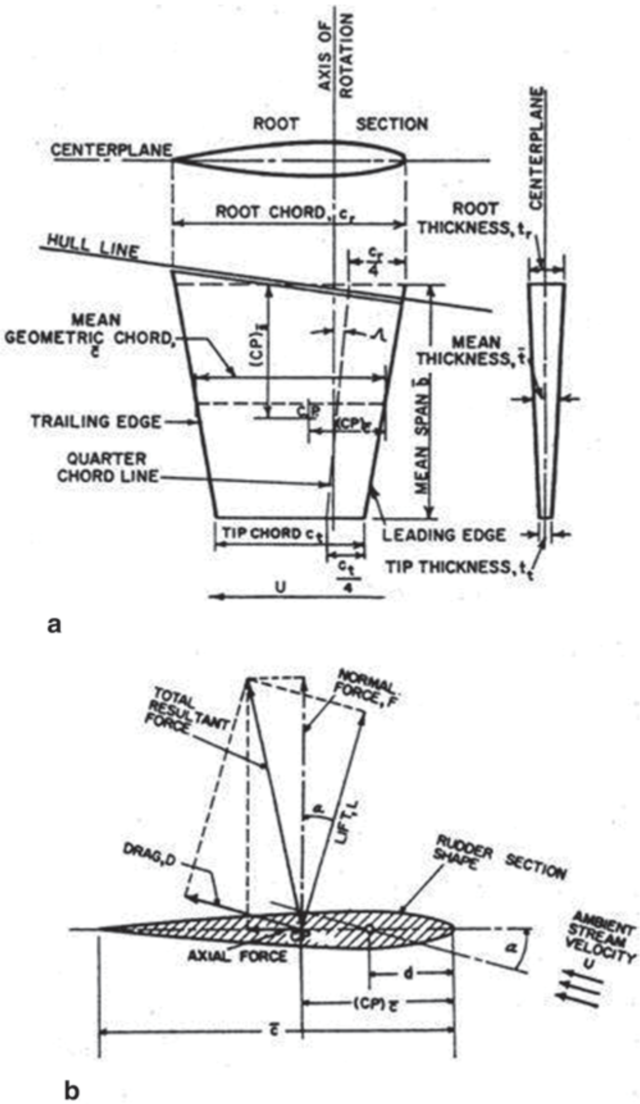


Fig. 5.34 **a** Typical dimensions of a rudder **b** Components of hydrodynamic rudder forces (Lewis 1988)

History The balanced rudder concept was first introduced by the ingenious British engineer *Isambard Kingdom Brunel* and used on the first ever built large iron ship, the SS Great Britain, launched in 1846 (Fig. 5.37). Its axis of rotation is behind its front edge. This means that when the rudder is turned, the hydrodynamic pressure acting upon the forward part of the rudder exerts a force which increases the angle of deflection, so counteracting the pressure acting on its rear part, which acts to

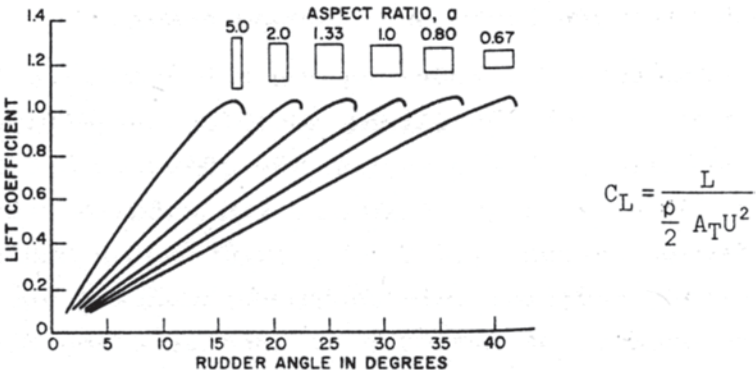


Fig. 5.35 Effect of aspect ratio on the lift coefficient C_L for various rudder angles α

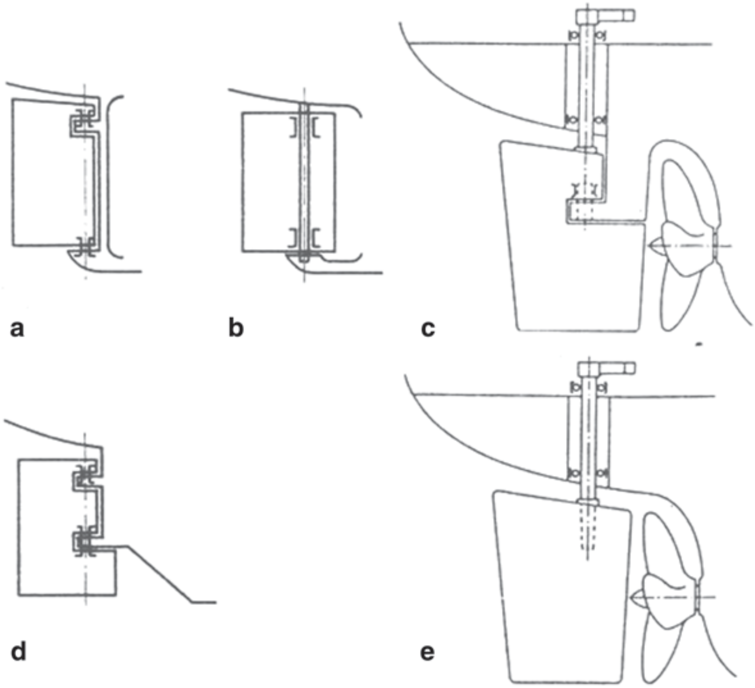


Fig. 5.36 Types of rudder. **a** Unbalanced rudder with upper and lower bearings. **b** Balanced rudder with bearings up and down. **c** Semi-balanced, half-hanging rudder with bearings (single screw). **d** Semi-balanced, half-hanging rudder for twin-screw ship. **e** Balanced, hanging rudder with upper bearing

Fig. 5.37 Rudder of SS Great Britain

reduce the angle of deflection. To avoid rudder instability, the area in front of the pivot is less than that behind. This allows the rudder to be moved with less effort than is necessary with an unbalanced rudder.

B. Criteria for Selecting the Position and Rudder Number

Position of rudder: The fitting of the rudder behind the accelerated propeller flow generates for the ship larger steering forces, which reach values of double of the magnitude of the corresponding forces for rudders outside of the propeller flow. In addition, for low speeds, the flow abaft of the propeller induces satisfactory steering forces for the ship, as opposed to rudders outside of the propeller flow.

The hull efficiency η_H is positively affected by the fitting of the rudder in the wake of the propeller flow and it is for single-screw ships about 1.0 and larger, while for twin-screw ships it is usually less than one. Certainly, the effect of the rudder on the coefficients that are determining η_H, namely the wake and thrust reduction coefficient, is very complex. In any case, it is considered that the rudder regains part of angular momentum of the water flow released behind the propeller and induces additional thrust forces, thus eventually it is reducing the required power to achieve a specified speed (increase of η_D).

Number of rudders: The application of a single rudder in the centreplane of the ship is constructionally the simplest solution. In this way, only one rudder mechanism is required and its fitting to ship is also simplified. If ship's draft does not allow the installation of one rudder with adequately large area (e.g. shallow water riverboats), more than one rudders are installed.

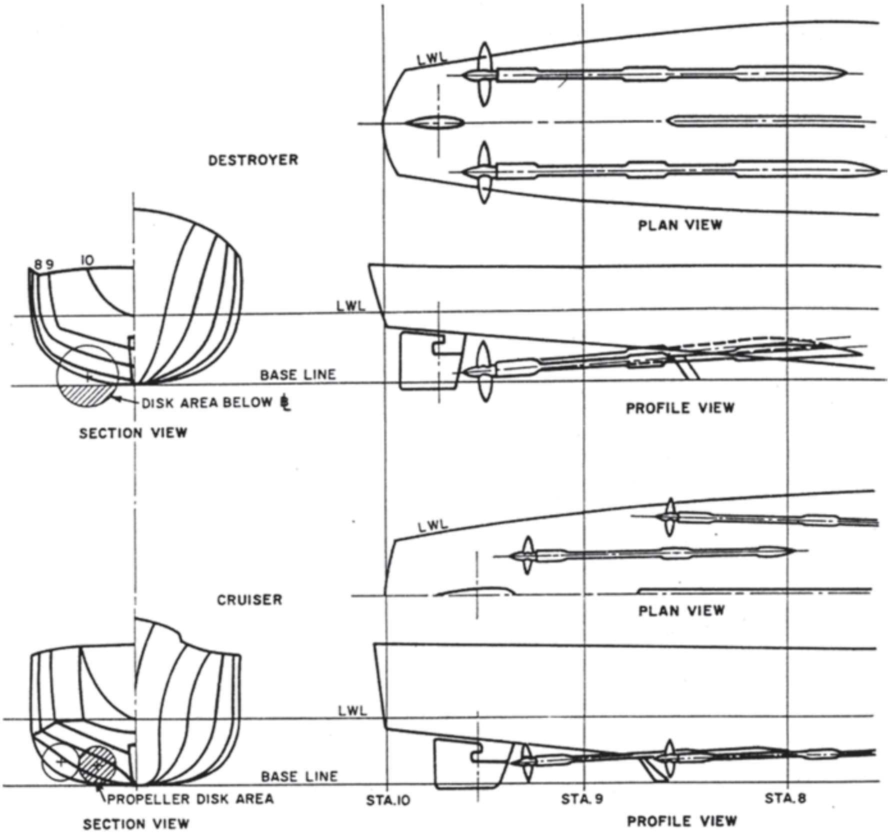


Fig. 5.38 Possible position of single rudder on multi-screw naval ships (Lewis 1988)

C. Applications

Single rudder behind a single propeller: It is the rule for practically all merchant ships. Attention should be paid to the pulling of the propeller end shaft for repairs, which should be not obstructed by the rudder (proves more problematic for CPPs and shafts) .

Single rudder behind two (four) propellers: It may be found on past large passenger ships (ocean liners) and naval ships (see Fig. 5.38).

Two rudders behind two propellers: It is common to all ships with special requirements for easy manoeuvring, e.g. RoPax ferries, trawler-fishing vessels, river ships and naval ships.

The position of the axes (rudder stocks) of side rudders is often slightly inclined sidewards, in the upper part. Also, lengthwise, the rudders are placed slightly outside the line of the propeller end shafts to facilitate their removal for repair.

Table 5.9 Typical values of rudder area coefficients for merchant ships according to Strohbusch (1971)

Ship type	$C = L \cdot T/A_R$
Cargo ships	65–75
Bulk carrier	70–75
Tankers	60–80
Short sea cargo ships	30–50
Reefers	45–60
Past ocean liner passenger ships	up to 85
Large passenger ships	55–70
Car ferries/RoPax	35–50
Trawlers	33–40
Open sea tugs	30–40

Two rudders behind three propellers: Sometimes applied to naval ships
Multiple rudders: Interconnected multi-rudder systems are applied to river ships (Henschke 1964).

D. Rudder Area Indicative values for the required rudder area A_R, in relation to the lateral plan projection of ship's wetted surface $\approx L \cdot T$, are given in Table 5.9:

E. Other and Alternative Rudder System Devices

- **Evolution of Stern Rudder:**

  Kort nozzle rudder (a)
  Pleuger active rudder (*propeller rudder*) (b)
  Azimuthal podded drive (c) (Figs. 5.39, 5.40, 5.41 and 5.42)

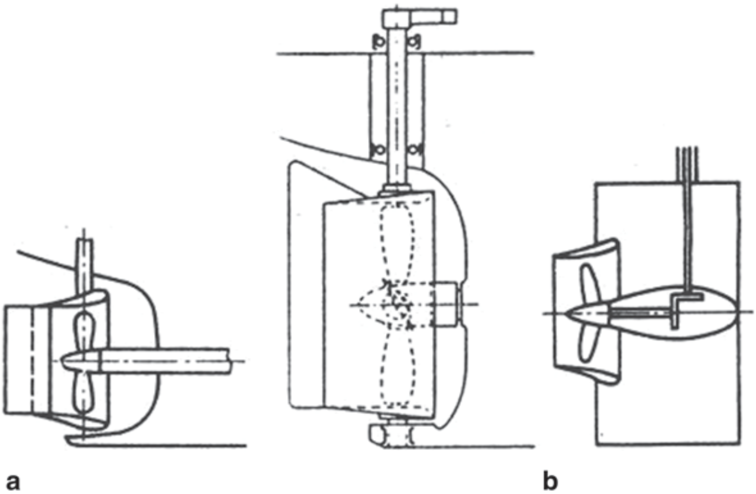


Fig. 5.39 **a** *Kort* nozzle rudder. **b** Pleuger active rudder

Fig. 5.40 *Kort* nozzle rudder

Fig. 5.41 *Pleuger* active rudder first fitted to *M/S Elisabeth Bowater* (1958)

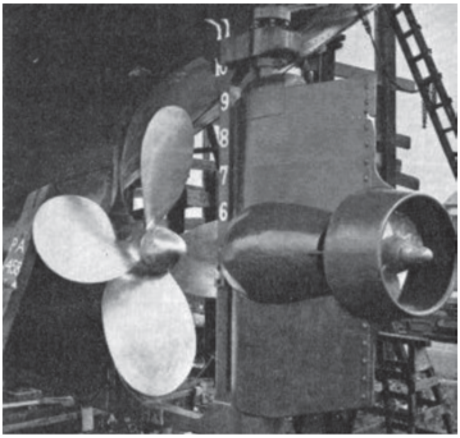
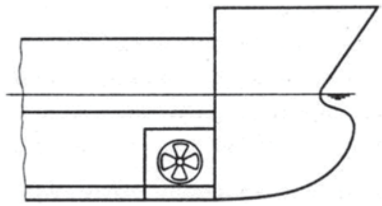


Fig. 5.42 Siemens–Schottel azimuth thrusters

Fig. 5.43 Bow thruster



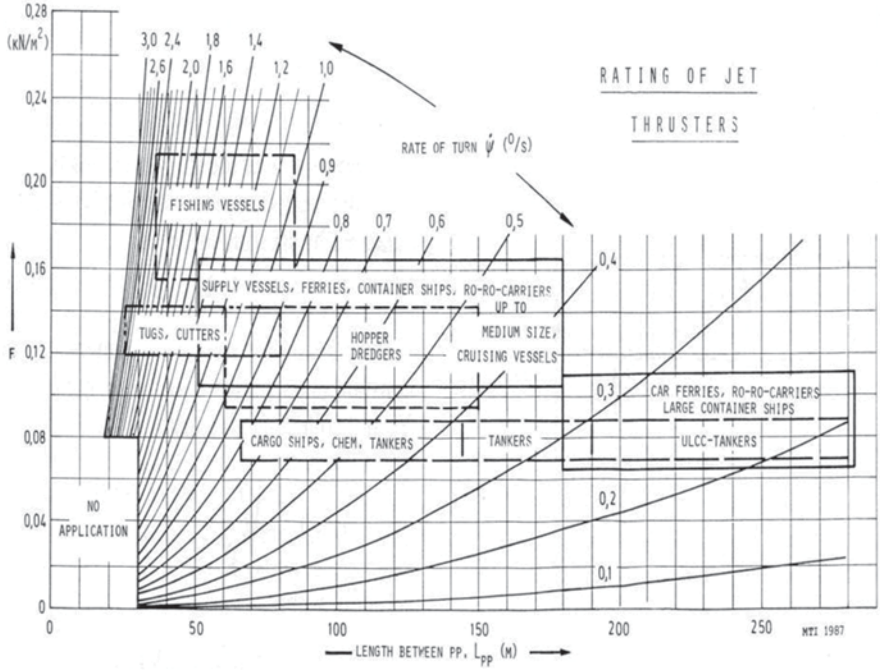


Fig. 5.44 Diagram for the selection of active bow thruster according to Brix (1993)

- **Bow Steering Devices:**
  - Regular type bow rudder for ships with special manoeuvring requirements (car ferries and pilot boats).
  - Active bow thrusters to facilitate manoeuvrability in limited waters, channels, ports etc. (car ferries, large passenger ships, large container ships etc.) (Figs. 5.43 and 5.44)

Lateral thrust: Y_0 (kN) = F (kN) · L_{PP} · T
Required power: P_0 (kW) = Y_0/C_0, where:
$C_0 \approx 0.150$ kN/kW (specific thrust coefficient for bow thruster; Fig. 5.45).

Fig. 5.45 Single and twin bow thrusters

The fitting of single and multiple *bow* thrusters is possible. For high manoeuvring performance ships (e.g. large passenger ships) , the fitting of active *stern* side thrusters is also common, unless the ships are fitted with Azimuthal podded drives.

F. Manoeuvrability

- **General:** Good manoeuvrability disposes a ship with the following characteristics:
  - Small turning diameter (small 'tactical diameter')
  - Rapid turning
  - Rapid equilibration of forces in the fully developed turning circle
- **Requirements on manoeuvrability:**
  - Interim international standards for a ship's manoeuvrability are laid down in IMO's Resolution A.751, adopted on November 4, 1993; they refer to a set of criteria for a ship's

    Turning ability: The advance should not exceed 4.5 ship lengths and the tactical diameter should not exceed 5 ship lengths in the turning circle manoeuvre.

    Initial turning ability: With the application of 10° rudder angle to port/starboard, the ship should not have travelled more than 2.5 ship lengths by the time the heading has changed by 10° from the original heading.

Yaw-checking and course-keeping ability: Control by conducted standard *zig-zag* tests

Stopping ability: The track reach in the full astern stopping should not exceed 15 ship lengths.

– For vessels with enhanced manoeuvring requirements, for instance, tugboats, a minimum turning tactical diameter is required, e.g. d_T/L up to $\cong 2.0$. The requirement for rapid turning is easily satisfied in practice due to their small size (small mass/inertia).
– For all common type merchant ships, 'rapid turning' is required for safety reasons against collision, which is controlled by the ship undergoing standardized *zig-zag* tests; for the turning tactical diameter we have mostly $d_T/L \cong 3.5-5.0$.

- **Effect of design parameters on manoeuvrability:**

a. **Turning moment** (see Sect. 5.3, A): The turning moment can be increased by increasing the rudder area A_R and the placement of the rudder in the wake of the propeller flow (increase of V_{Ru}). In addition, the rudder should be positioned as sternward as possible (increase of the lever arm a).
b. **Mass moment of inertia:** A ship's mass moment of inertia can be reduced by reducing the weights and optimizing their distribution, though this is difficult in practice.
c. **Hull form:** Sharp hull forms (large L/B) generally slow down the prompt turning. In particular, with respect to the lateral plan projection of the underwater hull surface, the fitting of a stern deadwood is enhancing the course-keeping ability of the ship, but adversely affects her turning ability. The same applies to vessels with stern trim. Generally, dominant V-type sections adversely affect the manoeuvrability of a vessel due to the relative increase of the projected underwater area.

G. Course Stability

- **General**: The course keeping ability/stability of a ship expresses ship's capability to hold her course without continuous corrective actions by the rudder.
- **Requirements**: All ships should have adequate course keeping stability for the following reasons:

– Reduced loading on the rudder bearings and driving mechanism,
– Fuel saving,
– Safe navigation in limited waters.

- **Effect of design parameters:**

a. **Mass and mass moment of inertia:** Generally, ships of large mass and large mass moment of inertia are difficult to be distracted from their course keeping, but also difficult to return to the original course, once deviating.

b. **Propeller flow:** The accelerated water flow due to the presence of the propeller generally stabilizes the course keeping of the ship, especially through the rudder that is located abaft, in the wake of the propeller.

c. **Hull form:** In general, those hull form characteristics that favour a ship's manoeuvring capabilities act adversely on her course-keeping ability. Thus, fine-lined, narrow ships (large *L*/*B*) favour the ship's course-keeping ability. Also, relatively sharp stern sections (of type V), with large draft close to the propeller and ending to a deadwood (like on upright keel on tugboats and fishing vessels), enhance a ship's course stability. Finally, a relatively large rudder (large area A_R) positively affects the constant course-keeping of the ship, in addition to the offered benefits of good manoeuvrability (Figs. 5.46, 5.47 and 5.48).

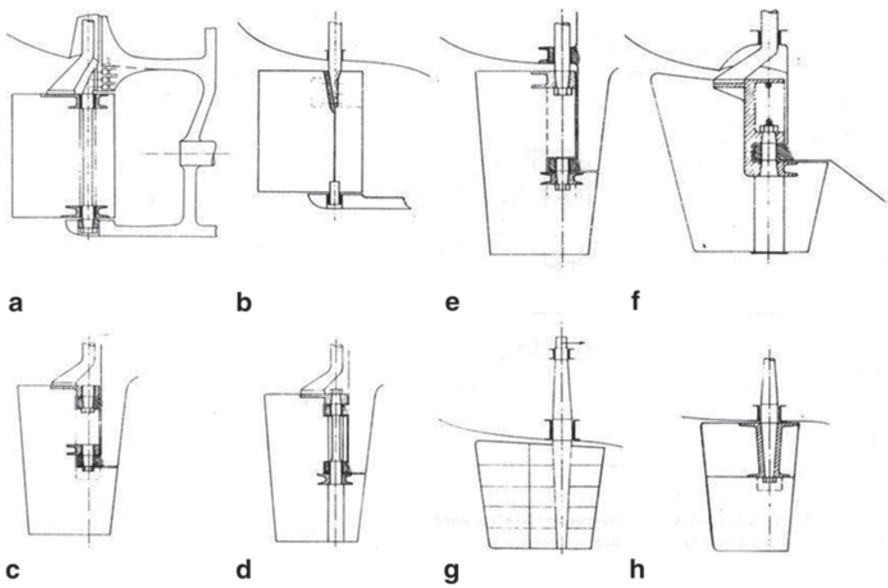


Fig. 5.46 Constructional examples of various types of rudders. **a** Old type balanced-rudder Simplex. **b** Similar to **a,** for small short sea ships. **c, d** Semi-balanced, half-hanging, new type Simplex. **e, f** Semi-balanced, half-hanging, new type. **g, h** Balanced/spade type, fully hanging, new type

Fig. 5.47 Example of bearing of fully hanging rudder by FAG-Kugelfischer (Schiff and Hafen 1986)

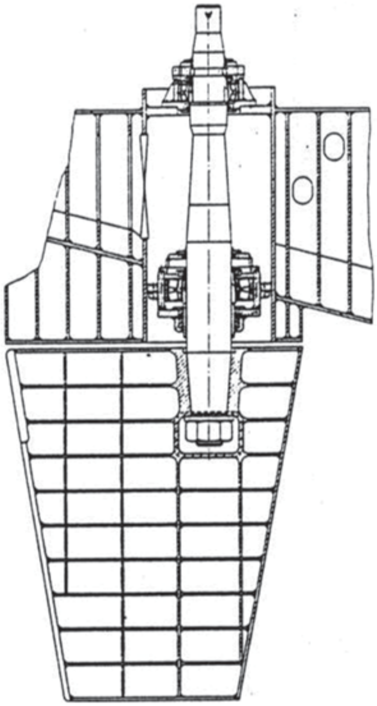
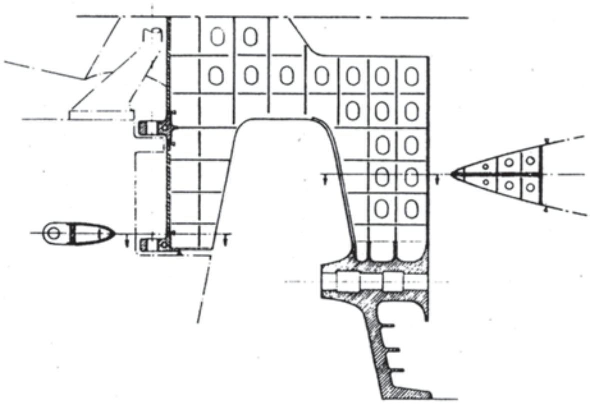


Fig. 5.48 Example of stern construction for semi-balanced, half-hanging rudder



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